# A Study of a Miniature In-Line Pulse Tube Cryocooler

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## ABSTRACT

A miniature pulse tube cryocooler has been designed and tested in our laboratory. The main part of the cryocooler, consisting of an in-line assembly of aftercooler, regenerator, cold heat exchanger, buffer tube and hot heat exchanger has an overall length of 31 mm. The regenerator is filled with SS #635 mesh screen and has a length of 12 mm. Despite the miniature dimensions, all the aforementioned components of the cryocooler are connected by pressure-tight joints allowing easy disassembly and are easily replaceable. The modular arrangement allows separate testing of the components and additional experimental tuning. The oscillating pressure is supplied by an external Eydan® compressor, and the entire assembly is fitted with an inertance tube and reservoir. A theoretical SAGE® model estimated 305 mW of cooling at 110 K for operation at 100 Hz with a filling pressure of 40 bars and a pressure ratio of 1.3 at the aftercooler.

As a result of the small dimensions, the actual performance of the cryocooler is strongly dependent on the uniform passage of gas between the cooler components, and in most cases differs from the theoretical assumptions. Nevertheless, experimental tuning of the components design and their interconnections in the cryocooler allowed us to reach the theoretical performance, and furthermore, to reach a higher cooling power than predicted by the SAGE® model. The actual cryocooler provides 400 mW of cooling at 110 K while operating at 103 Hz with the aforementioned fill pressure and pressure ratio. The cooler achieves a no-load temperature of 99 K, while the heat rejection temperature stabilizes at 305 K.

## INTRODUCTION

Miniaturizing a regenerative cryocooler has proven to be quite a difficult task. It is not sufficient to simply scale down the mechanical dimensions of the regenerative cooler and retain the same operating parameters. As a result of scaling the geometries down, the principal governing thermodynamic parameters of the problem are altered. The optimal cycle parameters for larger cooler dimensions are no longer valid at smaller characteristic lengths [1].

Increasing the fill pressure and the operating frequency are considered the key for miniaturization, as they compensate for the small volume of the working fluid [2]. However, losses associated with high speed flow and irreversible heat transfer, increase together with the frequency and smaller

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dimensions of a cryocooler. Thus, the higher frequency is beneficial as long as it does not escalate the irreversible processes in the cryocooler excessively. On the other hand, elevation of the filling pressure (with a constant pressure ratio) is limited, as it overloads the compressor, harming the cooling efficiency, and demands more bulky design of the cryocooler. Therefore, it was decided in this study to restrict both the frequency and fill pressure of the cryocooler to 100 Hz and 40 bar, respectively, while placing an emphasis on the reduction of losses and, therefore, enhancement of the total efficiency of the cryocooler.

To date, the most successful miniature pulse tube cryocooler developed in our laboratory remains the Minitech model MT*f* developed as part of the Ph.D. thesis of Isaac Garaway [3,4]. The total length of the assembled MT*f* (after cooler – regenerator – cold heat exchanger – buffer tube – hot heat exchanger) was 34mm, which included a 12mm long buffer tube and a 12mm regenerator filled with #635 S.S. mesh screen. Experimental results of MT*f* were obtained in a consistent and repeatable manner with sustained Tcold ~ 146K at 128 Hz for a considerable period of time. It provided 100mW of cooling power at Tcold ~ 160K ($\eta \sim 2\%$ of Carnot). The present work was initially intended to improve the aforementioned performance by means of further miniaturization, reduction of the low temperature, additional cooling power, and increase of the efficiency. For this purpose, a more detailed and accurate theoretical model (in comparison with the previous work) of the pulse tube was designed, studied in detail, and optimized using mainly the SAGE® software. Additionally, a new SAGE® model took into account the following design modifications of a new cryocooler relative to the MT*f*:

- Thin-wall stainless steel tubes replaced the PEEK tubes of both regenerator and buffer tube. This helped prevent leaks and, furthermore, made it possible to create a modular construction of the cryocooler. A clear disadvantage of the metal tubes instead of the plastic is the increased axial heat conduction. However, the heat conduction losses along the tubes are still minor compared to the internal losses of the regenerator and the buffer tube.
- All the components of the cryocooler are connected by pressure-tight joints allowing easy disassembly and are easily replaceable. The modular arrangement allows separate testing of the components, additional experimental tuning, and optimization of the tested components and the entire cryocooler.
- The inertance tube has a double in-series configuration, in which relatively thick and long inertance tube follows a thin and short one connected to the buffer tube. This arrangement allows obtaining of a smaller flow-to-pressure phase angle.
- Copper mesh screen stacks in the heat exchangers (aftercooler, cold and hot ends) are soldered to the inner surface of the exchangers in order to improve the radial heat transfer.
- A new vacuum chamber is designed for a higher vacuum, for fast and easy experiment preparation, and for a clear visible tracking of the operating cryocooler through the embedded window.

Throughout this discussion, the new pulse tube cryocooler will be referred to as the MTS*a* pulse tube, which is an acronym for Miniature Technion Stainless version *a*.

## PULSE TUBE MODELING IN SAGE®

### Overview

Perhaps the most popular cryocooler design software is SAGE®, a commercial package developed by Gedeon Associates [5]. It is a direct descendant of the GLIMPS® software, which was used widely within the Stirling industry for nearly ten years earlier. SAGE® introduced a drag-and-drop visual interface where the user could assemble a complete machine from standard components, such as pistons, cylinders, heat exchangers, etc. SAGE® also introduced an interactive optimization capability built into the visual interface. This has proven to be a very useful industry tool for cryocooler design. The calculations used in this program are based upon some theoretical models along

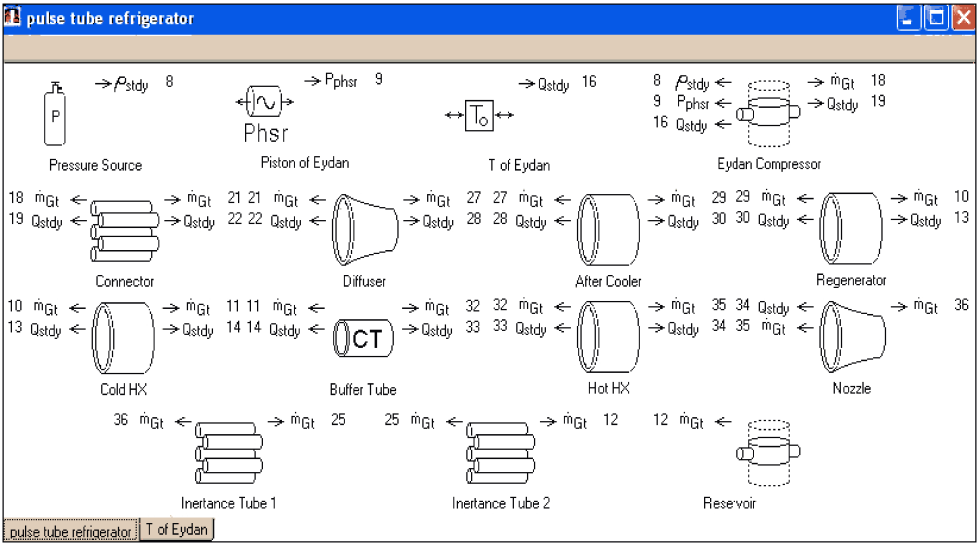


**Figure 1.** Root-level editing window of SAGE with the final model of the pulse tube cryocooler.

with some empirical correlations. The modeling is done in module form with model components organized logically in a hierarchical tree structure.

Figure 1 shows the root-level editing window of SAGE® with the final model of the pulse tube cryocooler. According to the figure the pulse tube model consists of 12 main components: Compressor (Eydan), Connector, Diffuser, Aftercooler, Regenerator, Cold Heat Exchanger, Buffer Tube, Hot Heat Exchanger, Nozzle, Inertance Tube 1, Inertance Tube 2 and Reservoir. Three additional components on the left upper side of the editing window serve to supply to the compressor a filling pressure, an oscillating displacement of the gas boundary, and a temperature of the compressor's housing. Each root-model component of the pulse tube contains a number of subcomponents (children) representing filling gas, tube walls, heat exchangers, and the like. These subcomponents themselves contain sub-subcomponents, such as gas inlets, heat-flow ends, conductors, and so forth. Root-level and subcomponents have boundary interconnections, which are the abstractions by which quantities such as gas flow, heat flux, etc., pass from one model component to another. Boundary connections among components are indicated graphically by matching numbered arrows attached to the individual model components. In this way it is possible to understand the physical connections among components.

## Optimization

Most of the relevant parameters such as dimensions, frequency, piston displacement, lattice parameters, temperatures etc. are determined numerically and can be iteratively optimized. There also exists quite a bit of flexibility with respect to user defined inputs, giving the designer control over many parameters such as material characteristics and operating boundary conditions. In the MTS*a* pulse tube some parameters of the cryocooler are set to be constant over the optimization for the benefit of the overall miniaturization, or because of the design restrictions. Thus, the regenerator is set to have 12 mm length, porosity 0.61 and wire diameter 20 microns, that correspond to a somewhat pressed stack of 300 pieces of #635 S.S. woven lattices, which fill the actual regenerator. The compressor is constrained to supply constant gas amplitude, producing pressure ratio about 1.3, or PV power about 12 W. The compressor provides the pressure oscillation to the cooler through the "Connector" component, which is 70 mm long. The temperature of the compressor's housing is set to 330 K, both aftercooler and hot heat exchanger are maintained at 300 K, and the cold end temperature is set to 100 K. Heat exchangers (aftercooler, cold and hot ends) possess #200, 50 micron copper lattice stack embedded in copper tubes. The filling pressure in the cryocooler is 40 bar,

**Table 1.** Final parameters of the pulse tube model.

| | Conn-ector | After Cooler | Regen-erator | Cold HX | Buffer Tube | Hot HX | Inert. Tube#1 | Inert. Tube#2 | Reser-vior |
|---|---|---|---|---|---|---|---|---|---|
| **Length [mm]** | **70** | **2** | **12** | **0.4** | **14** | **0.6** | **78** | **1881** | **50** |
| **Inner Diameter [mm]** | **1.1** | **4** | **3.8** | **4** | **3.8** | **4** | **0.38** | **1.1** | **16** |

and the operating frequency is 100 Hz. A coolant gas is ideal Helium, though specific heat, viscosity, and conductivity constants are temperature dependent, and taken from the SAGE® tables.

Optimized variables are diameters of all the components (excluding the compressor), lengths of the heat exchangers, of the buffer tube and of both inertance tubes. The volume of the reservoir has no optimum. A larger volume provides more uniform pressure in the reservoir, causing better performance of the cryocooler. The final volume of the reservoir was fixed at 10 cc, since a larger volume produced negligible effect on the cryocooler. "Diffuser" and "Nozzle" components, which appear as simple hollow cones, do not participate in the optimization of the cryocooler. From the point of view of the SAGE® model these are negative components, since they do not have any thermodynamic function, and merely add a dead volume. However, practically, both components are necessary for smooth and uniform passage of the gas through rapid change of tube sections.

The optimization is based on maximizing the net cooling power at the cold end of the pulse tube model, while all the ambient temperatures, as well as the piston stroke and frequency are kept constant. The entire pulse tube model was solved recurrently for finding the best combination of diameters and lengths of the model components. Afterwards, the optimum diameters of components such as inertance or buffer tube were rounded-off to the commercially available pipe sizes, and the optimization was completed by fitting the appropriate lengths. The final parameters of the MTS*a* pulse tube model are listed in Table 1.

## Results

SAGE® solves a model for one steady-state cycle assuming the unknown variables to have the form of Fourier series with a manually specified number of harmonics. The output is presented in the form of a list of vectors and scalars corresponding to the Fourier series amplitudes, phases and several integrations for various variables of each component and subcomponent of the model. For better understanding of the cryocooler behavior it was very useful to write a MATLAB® code reading the output file of the SAGE® model and converting the tabular results into graphical. Thus, flow and pressure oscillations in the heat exchangers of the final pulse tube model are shown graphically in Figure 2 according to the SAGE® solution for seven Fourier harmonics.

It is well known that the ideal regenerator must operate at constant entropy flow, so that exergy or work availability produced in the compressor should be transferred completely to the cold heat exchanger, where it is partially utilized as cooling power. Practically, both pressure and flow amplitudes decrease while passing the regenerator, and, in addition, a zero phase angle between them is not achievable by employing an inertance tube phase shifting mechanism. According to Fig. 2 the pressure amplitude is reduced by 18% in the regenerator, while flow amplitude decreases by 70%. In the buffer tube there is no pressure drop, thus the pressure curves of both cold and hot heat exchangers are identical; however, the flow curve is shifted back by 49 degrees due to the inertance tubes. As a result, the flow-to-pressure phase angle along the regenerator is reduced to between 28 and 22 degrees, while in the hot heat exchanger the angle is reduced to -21 degrees.

According to the bottom plot of Figure 2, despite the aforementioned somewhat large pressure and flow losses followed by non-ideality in the phase shifting, the cryocooler model still provides about 50 mW average cooling power at 100 K in the cold heat exchanger. Noteworthy is the fact that the temperature of the cold end gas varies between 96 and 104 K approximately, causing the

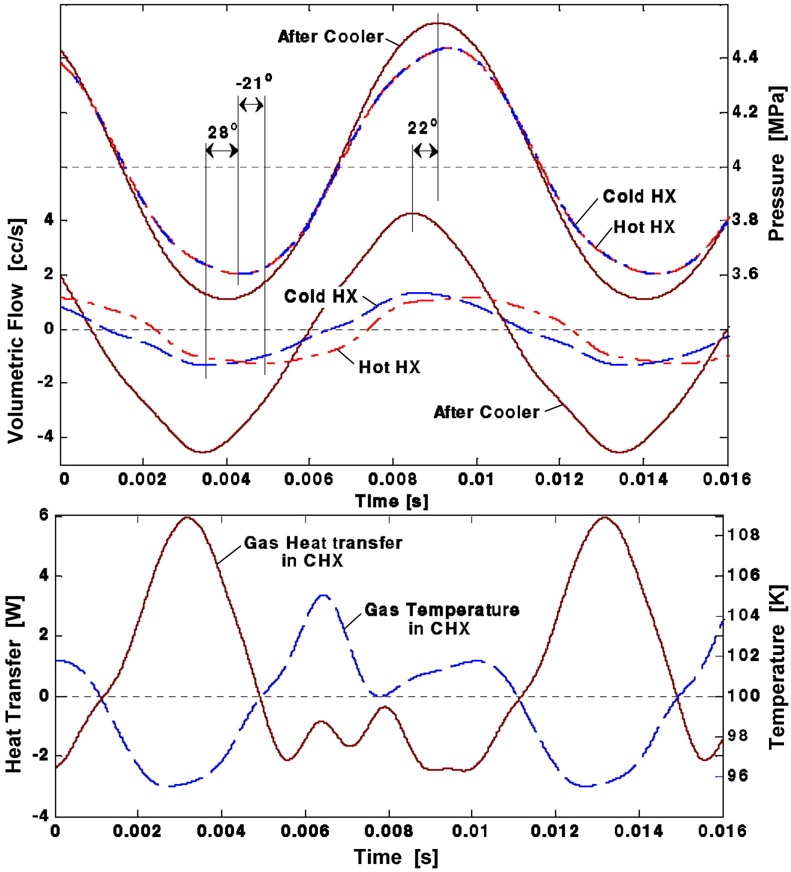


**Figure 2.** **Top** – volumetric flow versus pressure oscillation in the heat exchangers of the pulse tube model according to the SAGE solution. **Bottom** – gas heat transfer versus gas temperature in the cold heat exchanger.

heat to flow in both directions over the cycle. As a result, the peak of the heat transfer reaches almost 6 W of cooling, while in the second half of the cycle there is heating reaching a peak of 2.5 W. The aforementioned net cooling power of 50 mW is actually the integrated value over the entire cycle. Increasing the cold heat exchanger temperature up to 110 K causes the cold end gas to oscillate around 108 K, producing obvious ability to pump more heat. As a result, the cryocooler provides 305 mW of cooling at 110 K, while the PV power produced in the compressor is 11.6 W. Therefore, according to the SAGE® model, the Carnot efficiency of the MTS*a* pulse tube cooler should be 4.5% for 110 K in the cold end and 300 K in both warm heat exchangers.

## EXPERIMENTAL SYSTEM

### Pulse Tube Components

Despite the miniature dimensions, all the components of the MTS*a* pulse tube cryocooler are designed to be connected by pressure-tight joints allowing easy disassembly and are easily replaceable. The modular arrangement allows separate testing of the components and additional experimental tuning, as necessary.

Both regenerator and buffer tube are constructed from a thin-wall stainless steel tube (0.076 mm wall thickness) with two flanges soldered at both sides (see Fig. 3). In order to minimize the cold end mass while being detachable, the cold end flanges have a triangular geometry. Cold heat

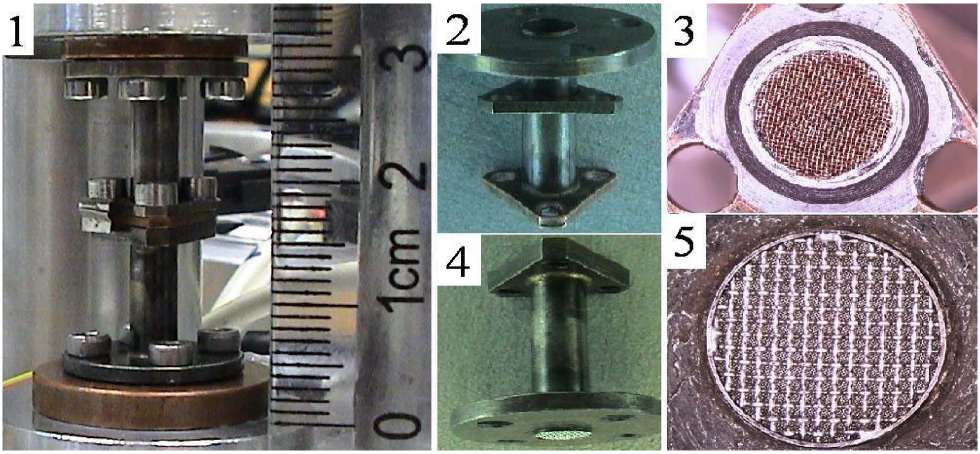


**Figure 3.** Main components of the Pulse Tube cryocooler. 1 - Assembly including after cooler, regenerator, cold heat exchanger, buffer tube and hot heat exchanger; 2 - Buffer tube; 3 - Cold heat exchanger embedded into the copper flange of the buffer tube and surrounded by lead sealing; 4 - Regenerator; 5 - Support mesh soldered to the tube at three locations holding the regenerator lattices.

exchanger lattices are embedded into the copper flange, which is soldered to the buffer tube and is covered by a stainless steel washer for better load distribution. The regenerator is filled with somewhat pressed 310 pieces of #635 S.S. mesh screens oriented at 45 degrees relative to each other, so that the axial flow passage can not be plugged by the adjoining wires. Both sides of the regenerator are supported by one layer of a brazed coarse mesh screen, which retain the regenerator lattices preloaded and fixed in place. Copper mesh stacks in the heat exchangers (aftercooler, cold and hot heat exchangers) are soldered to the inner surface of the exchangers in order to improve the radial heat transfer. Moreover, the soldering avoids possible vibrations and dislocations of the lattices.

According to the SAGE® model the hot heat exchanger possesses at optimum relatively small length, 0.6 mm, which is equivalent to a stack of six #200 mesh copper lattices. Certainly, in practice, six relatively coarse lattices are not enough to distribute uniformly the flow in the buffer tube with diameter of 3.8 mm, which expands from the inertance tube with diameter of 0.38 mm. SAGE® does not consider the transit phenomena between the model components, and actually assumes perfect flow passages. Practically, gas flowing from the inertance tube into the buffer tube creates a jet phenomenon, which causes an intense parasitic turbulence in the buffer tube and strongly harms the performance of the miniature cryocooler. The conventional way to prevent the jet phenomenon is to increase the number of lattices in the hot heat exchanger, which straighten the flow in a natural way. This method evidently enlarges the dead volume and shifts the cryocooler from the optimum operating point. Instead, we decided to retain the optimum number of the hot end lattices, and to eliminate the jet by adding a brake-jet element, which in fact is a solid slab, located in the "Nozzle" component (see Fig. 1) facing to the inertance tube aperture. The inertance tube jet collides with the slab and is spread in the radial direction. The warm end lattices merely complete the straightening of the flow. Incidentally, we tested both aforementioned methods of flow straightening in the same cryocooler, and concluded that employing the brake-jet element is evidently superior.

## Experiment Setup

Accurate analysis of the pulse tube cryocooler demands a series of measurements in real time during the operation. Pressure oscillations are measured by three threaded gauges at three locations: connector of compressor – aftercooler interface, hot heat exchanger – inertance tube interface, and reservoir (see Fig. 4). Temperature at the cold end is measured by a diode gauge with an accuracy of 0.1 K. Temperatures at both the aftercooler and hot heat exchanger are measured by K-Type thermocouples, since accuracy in these components is not as crucial. The cold end is equipped by a

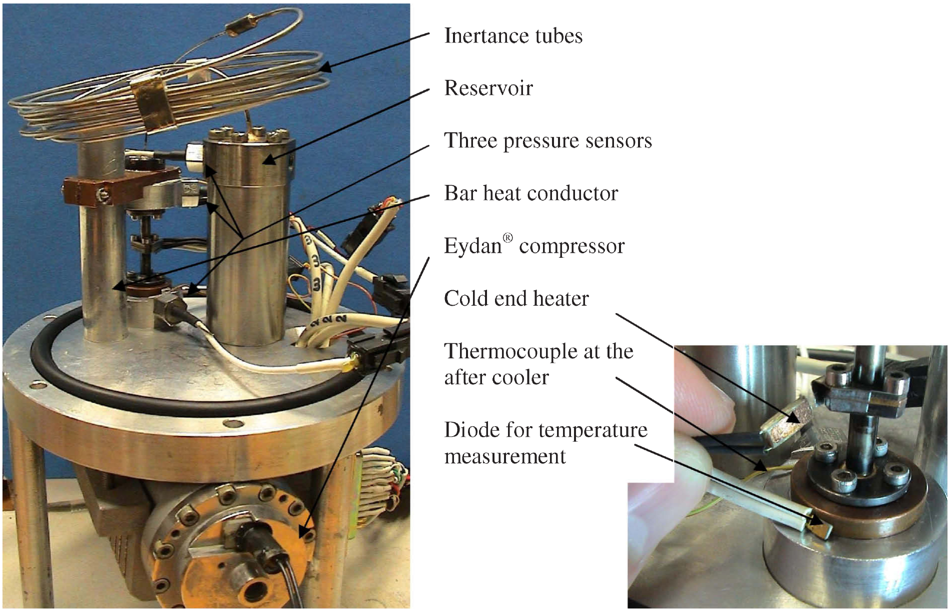


**Figure 4.** Central components of the experimental setup of the MTS*a* Pulse Tube cryocooler.

custom-made heater, which is attached by a springy clip. The diode temperature gauge is attached in the same manner on the side of the cold end assembly adjoining to the heater. The heater consists of a copper block with dimensions $8\times3.5\times1.5$ mm, which includes an embedded 32 AWG Nichrome wire retained by low temperature conductive epoxy.

Most of the pulse tube components, as well as the gauges and heater, are surrounded by vacuum environment. In order to maintain the vacuum level as high as possible, all the aforementioned electronic equipment are fed through a special vacuum feedthrough. The vacuum environment also demands that some consideration be made for transfering the heat out of the chamber from the hot heat exchangers. In our experimental setup there is a thick copper clip, which creates a thermal short between the hot end of the cryocooler and an aluminum rod, attached to the base of the vacuum chamber. The copper clip is designed to slide on the aluminum rod in order to fit an appropriate height. Heat from the after cooler is conducted directly to the vacuum chamber base. Actually, the chamber base serves as a kind of radiator for both warm heat exchangers of the cryocooler. Moreover, since the compressor is located outside the vacuum chamber below the base, it is reasonable to use the chamber as the heat sink for the entire cooler system. From a practical point of view, natural convection over the vacuum chamber surfaces is quite enough to maintain the system at an adequate temperature.

## Results

The Varian™ vacuum system employed in our experimental setup is able to create a vacuum of 10 torr. In practice, while being connected to our chamber containing the pulse tube cryocooler, the vacuum reaches a level of around 10 torr, which is quite enough to prevent a convection load on the cold end. In order to minimize the radiation heat transfer, the cold end was wrapped in a double layer of aluminized Mylar. As a result, the Mylar shield enabled the cold end no-load temperature of the MTS*a* cryocooler to decrease by an additional 2-3 degrees.

Figure 5 shows the cooldown process and the achieved no-load temperatures of the MTS*a* pulse tube cryocooler for the case of the highest vacuum, cold end wrapped in Mylar, and running at the optimum operating frequency of 103 Hz. Operation starts when the entire system is at a uniform room temperature of 298 K. During the operation, the cold end temperature drops down to

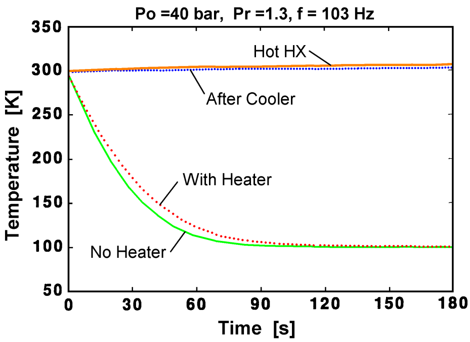


**Figure 5.** Cooldown process.

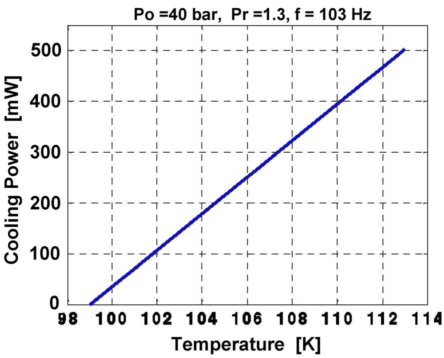


**Figure 6.** Steady state cooling capacity.

nearly 99 K (with or without a heater attached), the after cooler temperature stabilizes at 303 K, and the hot heat exchanger reaches nearly 307 K at steady state. The temperature in the hot heat exchanger is a bit higher, since the path from the hot end to the base of the vacuum chamber has a much larger heat transfer resistance than from the aftercooler, although the latter rejects a lot more heat during the operation.

Equipped with the heater, it takes 75 seconds of operation for the cold end to reach 110 K, which represents a steady state temperature for about 400 mW of cooling power (Fig. 6). With no heater attached, the cryocooler reaches 110 K after 64 seconds. Absolute steady state is obtained after 2-3 minutes of operation, depending on the cold end being equipped with or without the heater.

## CONCLUSIONS

The present research has proved that a temperature below 100 K at no-load, and several hundreds of mW cooling capacity at somewhat higher temperature, can be obtained by a miniature pulse tube cryocooler with a regenerator length of 12 mm, while operating at relatively low frequency of about 100 Hz, filling pressure of 40 bars, and pressure ratio of 1.3. Our MTS*a* pulse tube cryocooler practically achieved 99 K at no-load and provided 400 mW of cooling power at 110 K. Relatively large heat capacity and miniature cold end assembly provided fast cooldown. The aforementioned 400 mW at 110 K can be reached and maintained after 75 seconds of the cooldown process of MTS*a* pulse tube, while the load mass is about 0.4 gr (mass of the actual heater).

A fine agreement between the SAGE® model and the actual cryocooler results has been demonstrated. Differences in performance, e.g. cold temperature, cooling power and optimum frequency, are minor. In fact, the actual cryocooler provided somewhat better performance than was predicted from the calculations. The SAGE® model predicted 305 mW at 110 K instead of the measured 400 mW, while the actual aftercooler and hot heat exchanger rejected heat at 303 K and 307 K, respectively, instead of 300 K assumed in the SAGE® model. One may assume that maintaining the actual warm heat exchangers at the nominal 300 K should decrease the low temperature by an additional 3 K at least.

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